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SEA TRIALS OF THE DAMPED VIBRATION ABSORBER FITTED TO HMAS BRUN--ETC(U)

JAN 79 P FARRELL, A GOLDMAN

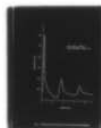
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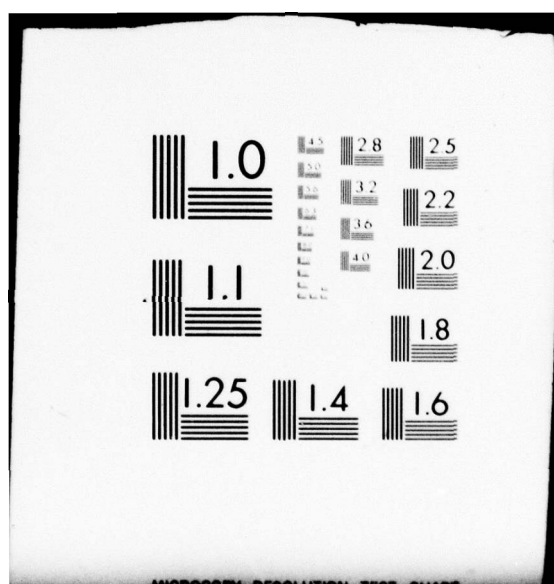
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Structures Technical Memorandum 286

SEA TRIALS OF THE DAMPED VIBRATION ABSORBER
FITTED TO IMAS BRUNEI.

P.A. FARRELL and A. GOLDMAN

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6 SEA TRIALS OF THE DAMPED VIBRATION ABSORBER
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10 P.A. FARRELL and A. GOLDMAN

11 Jan 79
12 16p.

14 ARL/STRUCT-M-286

SUMMARY

↓ A damped vibration absorber, similar to that previously installed in HMAS Balikpapan, has been fitted to HMAS Brunei. A series of tests, conducted on the latter ship, to determine the characteristics of the absorber and to provide an initial tuning guide, is described. ↗

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1. INTRODUCTION

The Landing Craft-Heavy (LCH), operated by the RAN, are subject to excessive vibration in the two-node vertical bending mode of the hull. This mode is excited by wave action and the modal damping is so low (typically 0.5% of critical), that these vibrations decay very slowly. To increase the damping, and hence decrease the vibration, a damped vibration absorber was designed (reference 1), and installed in HMAS Balikpapan, a typical LCH. After various trials (references 2 and 3), the operation of the absorber was considered to be satisfactory, so a second was constructed and installed in a sister ship, HMAS Brunei. Although the two ships are nominally the same, their characteristics vary slightly. The two absorbers cannot be considered identical and hence a series of trials was conducted on HMAS Brunei. These trials, although not as comprehensive as those conducted previously, were sufficient to determine the basic parameters for initial tuning of the absorber. The final recommended tuning curve will be determined after the ship has been in service with the absorber for some months and the data, collected over this period, analysed.

The trials consisted of two sets. First the ship was excited, with the ARL mechanical vibrator, in calm water. These steady state tests provided information on the natural frequency of both the absorber and the ship. Because of the power limitation of the mechanical vibrator, these tests involved only small vibration amplitudes. The second set of tests involved exciting the ship with wave action and analysing the resulting vibrations using the techniques described in reference 2. The amplitudes involved here are larger, and more accurately represent the vibration of the ship when in service.

The two sets of trials and the corresponding results are described in the following sections.

The units shown in the figures and table are those in which the ship's instruments (pressure gauges, draft gauges etc) are calibrated.

2. STEADY STATE TESTS

During these tests the ARL mechanical vibrator was used to excite the ship in still water of depth at least 18 metres (approx. 60 ft.). To determine the ship's natural frequency the absorber was locked and the frequency of excitation varied until the frequency at which the peak response occurred was determined. For a lightly damped structure, such as the ship, this frequency is also the natural frequency. This procedure was followed for the three loadings shown in Table 1 and the results plotted in Figure 1. The variation of natural frequency with loading is essentially linear over the range of loadings encountered in service. For comparison the corresponding curve for HMAS Balikpapan, taken from reference 2, is shown on the same figure.

The second stage of the steady state testing was to determine the natural frequency of the absorber as a function of air spring pressure. The power of the mechanical vibrator was insufficient to excite the absorber with all its dampers (shock absorbers) attached, so these were temporarily removed. The ship was excited as before but this time the absorber was free to oscillate. The vibrations of the ship and of the absorber relative to the ship were measured and the frequency of excitation varied until the ratio of these vibration amplitudes was a maximum. This frequency is the natural frequency of the absorber for that particular air spring pressure. These results are shown in Figure 2 where again the corresponding curve for Balikpapan (from ref. 2) is also plotted.

From these results it would be possible, using the equations of reference 1, to plot a graph of optimum absorber tuning versus ship's loading. However, these results were obtained from tests that essentially involved only small amplitudes of vibration of the ship and absorber. Since both the ship and absorber are non-linear, large amplitude tests are required and these are best done using wave action as the excitation. The results of the steady state tests were thus used to indicate the ranges over which the tests of the following section should be conducted.

3. TRANSIENT TESTS

For these tests the dampers were replaced on the absorber. It was intended that all 16 dampers be replaced but due to attachment difficulties only 13 were reattached. Two of the missing dampers were from the group furthest from the pivot, and the third was the port upper damper closest to the pivot.

The ship was then sailed into the waves to maximise the hull vibration and the response of the ship and of the absorber were recorded for analysis at ARL after completion of the tests. These tests were carried out for a series of air spring pressures and for the heavy and light ship conditions shown in Table 1.

Using the analysis methods described in reference 2, these transient tests yield the natural frequency of the ship for each of the two loading conditions, and also the natural frequency of the absorber for each of the pressures used. The two ship natural frequencies are plotted on Figure 1, where they are significantly below the steady state curve. This change in frequency is due to the non-linearity of the ship which causes the larger-amplitude transient test to indicate a lower natural frequency than the smaller-amplitude, steady state test. There was a similar lowering in the natural frequency of Balikpapan as determined from a transient test (reference 2) and this point is also plotted on Figure 1.

The natural frequency of the absorber for each pressure used in the transient tests is plotted in Figure 2. The curve derived from the transient tests (when 13 dampers were fitted to the absorber) is above the curve from the steady state tests (when no dampers were fitted). This increase in natural frequency is due to the presence of the dampers, as was previously noted in reference 2 (see Figure 5 of reference 2).

The optimum tuning occurs when the absorber frequency is approximately 98% of that of the ship. From the transient-derived curves of Figures 1 and 2, and applying the 98% relationship, an optimum tuning curve of absorber spring pressure versus ship's displacement was determined and is presented in Figure 3. This curve is intended as an interim guide only, and will be modified as more data from the ship become available.

These tests also gave an estimate of the damping of the absorber and with 13 dampers fitted they indicated that the damping ratio was about 11% to 12% of critical. The optimum damping is considered to be approximately 9% and hence the damping should be reduced.

From Figure 3, the optimum top spring pressure is 62 PSIG when the ship's loading is 380 tons and figure 4 shows the frequency spectrum of the ship acceleration measured for approximately these conditions. The peak near 3.6 Hz (the two-node, vertical hull bending mode) is essentially a single peak, whereas reference 1 indicates that, for correct tuning, there should be two peaks in this region (see, for example, Figure 17 of reference 2). The absence of the clearly defined twin peaks here is another indication that the damping is too high.

4. CONCLUSIONS

Initial trials of the damped absorber fitted to HMAS Brunei have been completed. From the data obtained a recommended tuning curve for the absorber has been produced. The tests indicate that the damping is too great, and it is recommended that this be reduced to improve the absorbers operation.

Upon completion of the tests, the Brunei was instrumented to provide data concerning the operation of the absorber during service. As these data become available, and are analysed, further recommendations concerning the tuning of the absorber will be made.

REFERENCES

1. G. Long and P.A. Farrell The design and installation of a damped vibration absorber.
A.R.L. Structures Note 440, August 1977.
2. G. Long and P.A. Farrell Sea trials of a damped vibration absorber on HMAS Balikpapan.
ARL Structures Tech. Memo. 270, January 1978.
3. P.A. Farrell and G. Long An evaluation of a damped vibration absorber in rough seas.
ARL Structures Tech. Memo. 280, September 1978.

TABLE 1. SHIPS LOADING DURING TESTS

SHIP CONDITION	HEAVY	LIGHT	MEDIUM
Design Lightship & BHD 28	320.2	320.2	320.2
Absorber Bottles and Panel	13	13	13
Crew and Effects	2	2	2
Stores and Misc. Equip.	4.3	4.3	4.3
Test Equip. and Team	2	2	2
Discrepancy Between Design & Actual	14	14	14
No. 2 Ballast P	34.2		
S	28.0		
b	5.1		20.2
No. 3 Void P			
S			
b			
No. 4 FWD Ballast P	1.4	-	28
S	7.1	-	25.5
No. 4 AFT Ballast P	1.2	-	-
S	-	-	-
No. 5 Ballast P	18	2.5	2.5
S	18	2.5	2.5
Bilge P	6.1	0.3	0.3
S	9.2	4.5	4.5
No. 4 FWD Fuel (extended range)	-	-	-
AFT Fuel	9.0	7.2	7.2
Daily Service	0.3	0.3	0.3
Lube Oil & Hydraulic Oil	1.8	1.8	1.8
No. 5 FW	1.1	4.0	4.0
CALCULATED WEIGHT	496.6 Tons	378.6 Tons	452.3 Tons
Crafts FWD P	3'-10 $\frac{1}{2}$ "	2'-6 $\frac{1}{2}$ "	
S	3'-10 $\frac{1}{2}$ "	2'-8 $\frac{1}{2}$ "	
Mean	3'-10 $\frac{3}{8}$ "	2'-7 $\frac{1}{2}$ "	
AFT P	7'-6"	6'-6 $\frac{1}{2}$ "	
S	7'-6 $\frac{1}{2}$ "	6'-8 $\frac{1}{2}$ "	
Mean	7'-6 $\frac{1}{2}$ "	6'-7 $\frac{1}{2}$ "	
Amidship Mean	5'-8 $\frac{5}{16}$ "	4'-7 $\frac{1}{2}$ "	
WL Below θ P	$\frac{1}{2}$ "	1'-2"	
S	0	1'-1"	
Mean	$\frac{1}{2}$ "	1'-1 $\frac{1}{2}$ "	
Mean Draft By θ = 5'-8 $\frac{3}{8}$ " -	5'-8 $\frac{1}{8}$ "	4'-6 $\frac{7}{8}$ "	
Mean B θ			
Trim Between Marks	3'-8 $\frac{7}{8}$ "	4'-0"	
Trim = Trim Between Marks -	1'-6 $\frac{5}{8}$ "	1'-9 $\frac{3}{4}$ "	
2'-2 $\frac{1}{2}$ "			
DISPLACEMENT FROM HYDROSTATICS	498 Tons	386 Tons	453 Tons

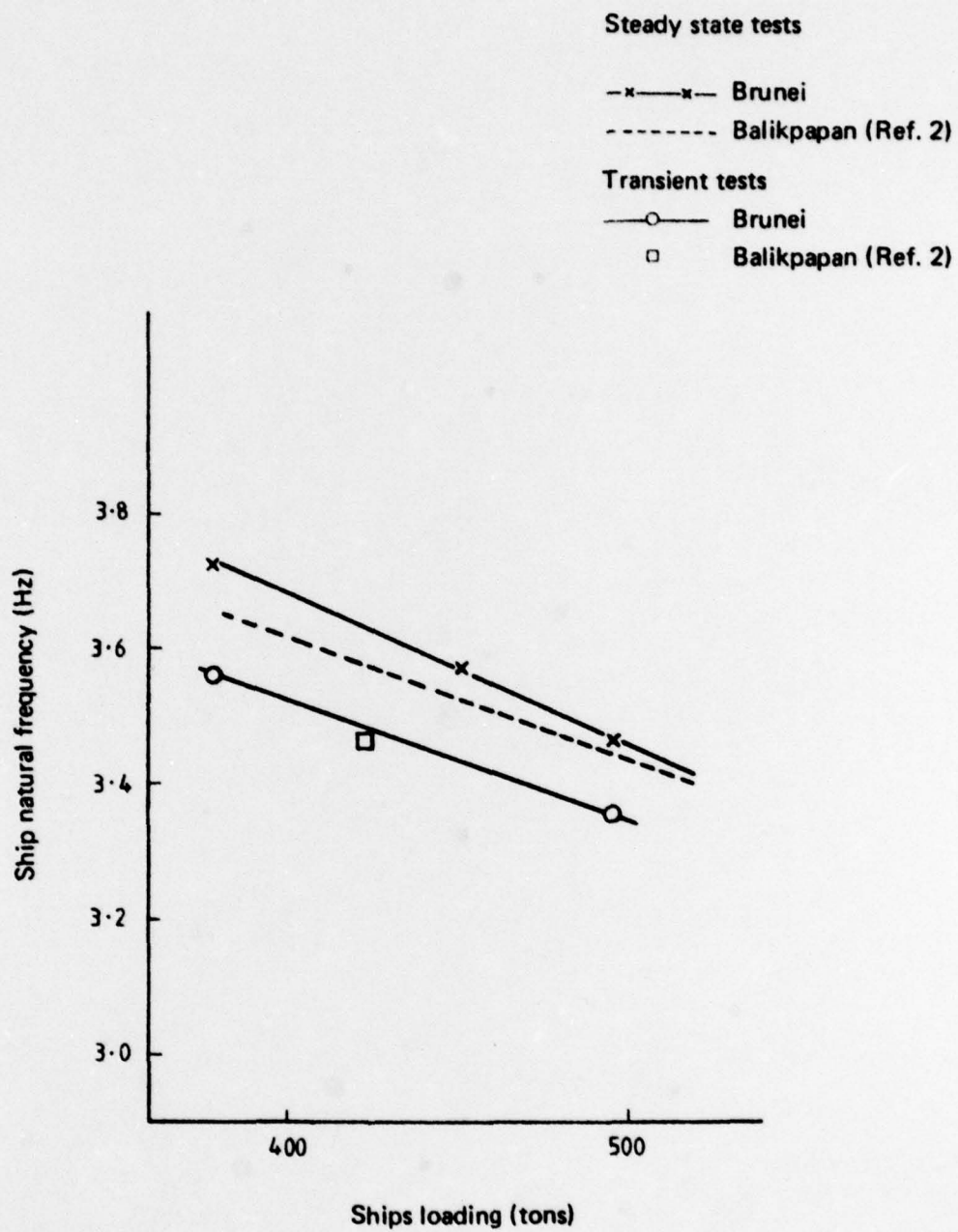


FIG. 1. VARIATION OF SHIP'S NATURAL FREQUENCY WITH LOADING

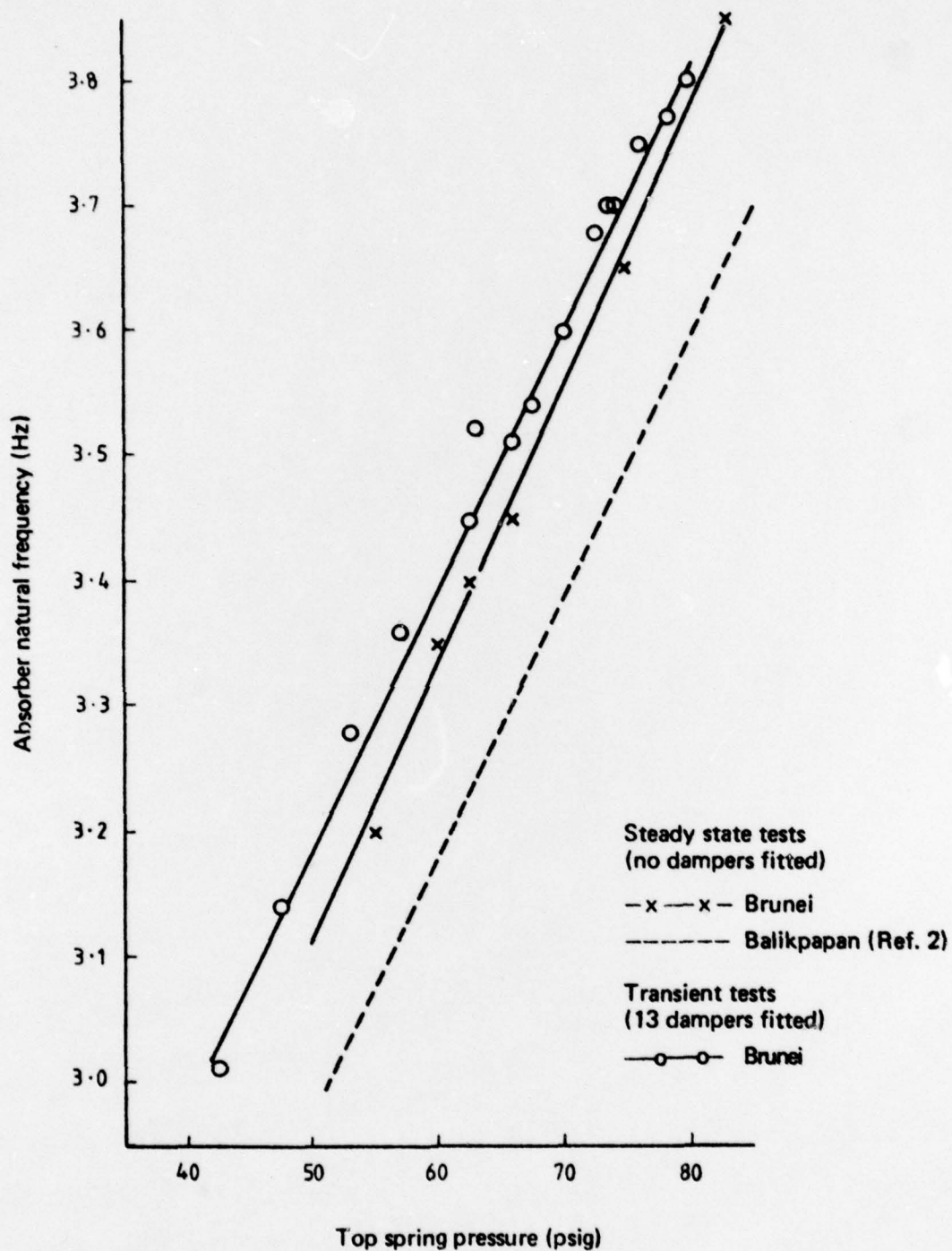


FIG. 2. VARIATION OF ABSORBER NATURAL FREQUENCY WITH TOP AIR SPRING PRESSURE

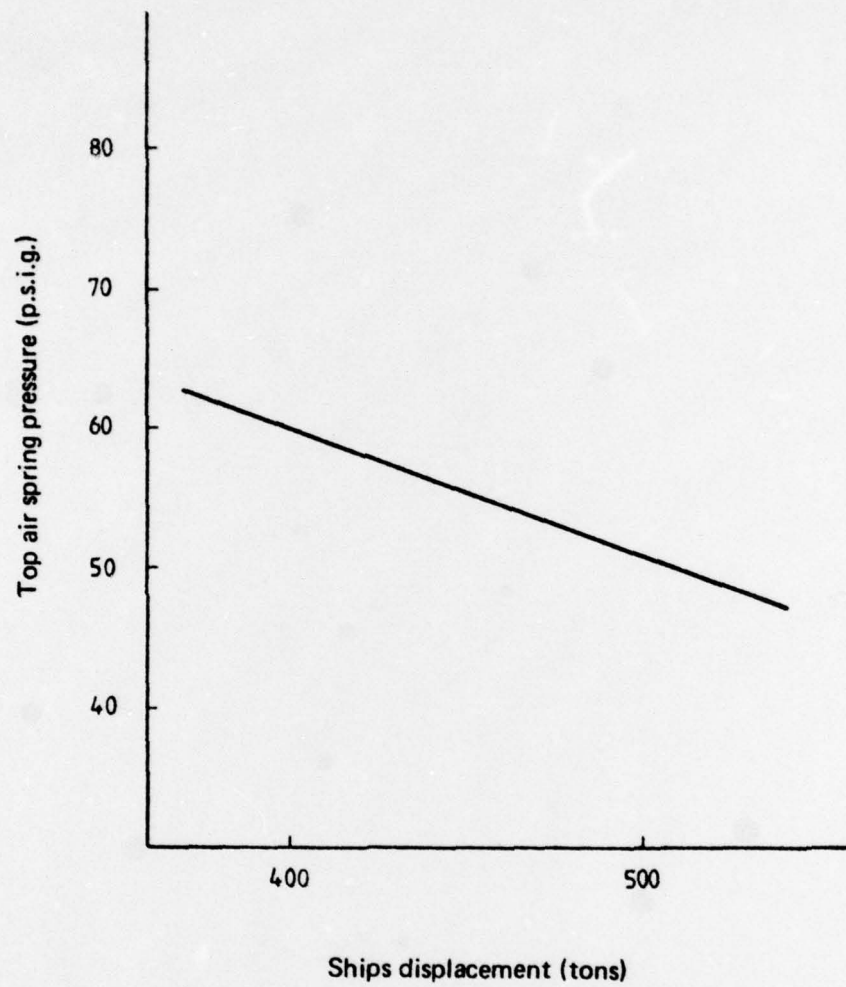


FIG. 3. RECOMMENDED TOP AIR SPRING PRESSURE

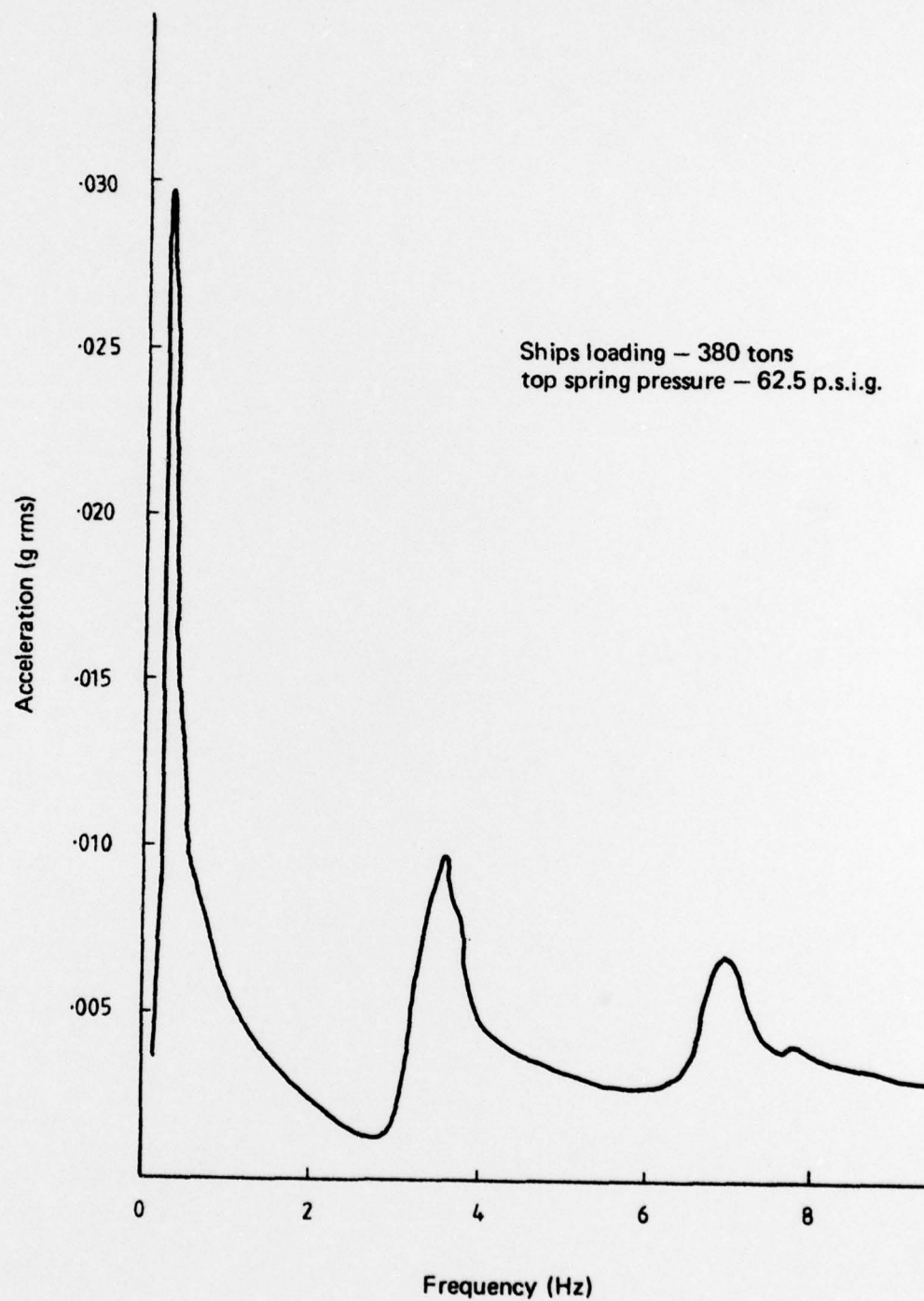


FIG. 4. FREQUENCY SPECTRUM OF SHIP ACCELERATION RESPONSE

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Security classification of this page: UNCLASSIFIED

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|--|---|
| <p>1. DOCUMENT NUMBERS</p> <p>a. AR Number:
AR-001-327</p> <p>b. Document Series
and Number:
Structures Technical
Memorandum 286.</p> <p>c. Report Number:
ARL-Struc-Tech-Memo-286</p> | <p>2. SECURITY CLASSIFICATION</p> <p>a. Complete document:
UNCLASSIFIED</p> <p>b. Title in isolation:
UNCLASSIFIED</p> <p>c. Summary in isolation:
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3. TITLE:

SEA TRIALS OF THE DAMPED VIBRATION ABSORBER
FITTED TO HMAS BRUNEI.

- | | |
|--|---|
| <p>4. PERSONAL AUTHORS:</p> <p>P.A. FARRELL
and
A. GOLDMAN</p> | <p>5. DOCUMENT DATE:
JANUARY, 1979</p> <p>6. TYPE OF REPORT AND PERIOD
COVERED:</p> |
| <p>7. CORPORATE AUTHOR(S):
Aeronautical Research Laboratories.</p> | <p>8. REFERENCE NUMBERS</p> <p>a. Task:
NAV 76/18</p> <p>b. Sponsoring Agency:</p> |
| <p>9. COST CODE:
232610</p> | |

- | | |
|--|---|
| <p>10. IMPRINT:
Aeronautical Research
Laboratories, Melbourne.</p> | <p>11. COMPUTER PROGRAM(S)
(Title and Language(s)):</p> |
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12. RELEASE LIMITATIONS (of the document)
Approved for Public Release.

12-0. OVERSEAS: NO ☐ P.R. ☐ 1 ☐ A ☐ B ☐ C ☐ D ☐ E ☐

13. ANNOUNCEMENT LIMITATIONS (of the information on this page):
No Limitation.

- | | |
|---|--|
| <p>14. DESCRIPTORS:</p> <p>Vibration Damping Tests</p> <p>Vibration Tests Landing Craft</p> <p>Vibration Damping Ships</p> | <p>15. COSATI CODES:</p> <p>2011</p> <p>1310</p> |
|---|--|

16. ABSTRACT:

A damped vibration absorber, similar to that previously installed in HMAS Balikpapan, has been fitted to HMAS Brunei. A series of tests, conducted on the latter ship, to determine the characteristics of the absorber and to provide an initial tuning guide, is described.

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